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Indian Standard

METHOD OF MEASUREMENT OF FLUID FLOW BY MEANS OF VENTURI METERS

PART II COMPRESSIBLE FLUIDS

(Second Reprint JANUARY 1998)

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NEW DELHI 110002

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December 1975

Indian Standard

METHODS OF MEASUREMENT OF FLUID FLOW BY MEANS OF VENTURI METERS

PART II COMPRESSIBLE FLUIDS

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(*Continued on page* 36)

Indian Standard

METHODS OF MEASUREMENT OF FLUID FLOW BY MEANS OF VENTURI METERS

PART II COMPRESSIBLE FLUIDS

0. FOREWORD

0.1 This Indian Standard (Part II) was adopted by the Indian Standards Institution on 5 February 1975, after the draft finalized by the Fluid Flow Measurement Sectional Committee had been approved by the Civil Engineering Division Council.

0.2 Measurement of gases flowing through pipes is encountered in several types of industries. The same type of orifice plates, nozzles and venturi meters used for incompressible fluids are also used for measurement of compressible fluids.

0.3 Venturi meters have advantages like small overall loss of head due to their streamlined structure, the relatively large volume of flow they can deal with and ease of installation in a pipeline. Further, measurements can be made over a wide range of velocities. However, the flow coefficient of venturi meter depends on several factors like the shape of contraction and divergent cones, range of velocities, roughness at the throat, etc.

0.4 In the formulation of this standard due weightage has been given to international co-ordination among the standards and practices prevailing in different countries in addition to relating it to the practices in the field in this country. This has been met by deriving assistance from the following publications:

ISO/R781-1968 Measurement of fluid flow by means of venturi tubes

BS 1042 (Part I) : 1964 Methods for the measurement of fluid flow in pipes, Part I Orifice plates, nozzles and venturi tubes

0.5 This standard is one of a series of Indian Standards on methods of measurement of fluid flow in closed conduits. Other standards published so far in the series are:

IS : 2951 (Part I)-1965 Recommendation for estimation of flow of liquids in closed conduits: Part I Head loss in straight pipes due to frictional resistance

IS : 2951 (Part II)-1965 Recommendation for estimation of flow of liquids in closed conduits: Part II Head loss in valves and fittings
IS : 2952 (Part I)-1964 Recommendation for methods of measurement of fluid flow by means of orifice plates and nozzles: Part I Incompressible fluids
IS : 2952 (Part II)-1975 Recommendation for methods of measurement of fluid flow by means of orifice plates and nozzles: Part II Compressible fluids
IS : 4477 (Part I)-1967 Methods of measurement of fluid flow by means of venturi meters: Part I Liquids

0.6 In reporting the results of a test or analysis made in accordance with this standard, if the final value, observed or calculated, is to be rounded off, it shall be done in accordance with IS : 2-1960*.

1. SCOPE

1.1 This standard (Part II) specifies the profile, sizes and material of construction of venturi meters and venturi nozzles and outlines the procedure for the measurement of flow of gases under steady conditions up to a Mach number of 0·5 at the throat.

1.2 Details of connections between primary and secondary elements are proposed to be covered in a separate standard.

1.3 This standard does not cover the measurement of steam, wet gases or mixtures.

1.4 This standard also provides the data on discharge coefficients applicable to the following conditions for use in computing the rate of flow through these devices of specified profile when direct calibrations have not been made on them, and provides an example for computation of flow (*see* Appendix A):

Classical Venturi Tubes

a) *Rough cast convergent*:
   1) Upstream Reynolds number between $2{\cdot}1 \times 10^5$ and $2{\cdot}1 \times 10^6$,
   2) Pipe diameter between 100 and 800 mm,
   3) Diameter ratio between 0·3 and 0·75, and
   4) Pressure ratio greater than 0·75.

b) *Machined convergent*:
   1) Upstream Reynolds number between $2{\cdot}1 \times 10^5$ and $2{\cdot}1 \times 10^6$,
   2) Pipe Diameter between 50 and 250 mm,
   3) Diameter ratio between 0·4 and 0·75, and
   4) Pressure ratio greater than 0·75.

---

*Rules for rounding off numerical values (*revised*).

c) *Rough welded sheet-iron convergent*:
   1) Upstream Reynolds number between $2 \cdot 1 \times 10^5$ and $2 \cdot 1 \times 10^6$,
   2) Pipe Diameter between 200 and 1 200 mm,
   3) Diameter ratio between 0·4 and 0·7, and
   4) Pressure ratio greater than 0·75.

d) *Venturi nozzles*:
   1) Upstream Reynolds number between $0 \cdot 75 \times 10^5$ and 2×10^6,
   2) Pipe diameter between 65 and 500 mm,
   3) Diameter ratio between 0·316 and 0·78, and
   4) Pressure ratio greater than 0·75.

1.5 Individual calibrations should be made when the profiles (*see* **7** and **8**) and installation conditions (*see* **5.3**) differ from this standard.

2. TERMINOLOGY

2.0 For the purpose of this standard, the following definitions shall apply.

2.1 Primary Elements — The devices introduced in the pipe are called primary elements which term includes the pressure taps (all other instruments or devices required for measurement are known as 'secondary devices').

2.1.1 *Orifice or Throat* — The term orifice or throat means the smallest cross-sectional opening of the primary element allowing the passage of the fluid through the element.

2.1.1.1 Orifice (throat) of standard primary element is always circular and co-axial with the pipeline.

2.1.2 Standard venturi meters are described as follows:

a) *Venturi Meter* — Venturi meter has a cylindrical portion called entrance cylinder (*A*) connected to a conical converging portion called the convergent (*B*), a cylindrical part called the throat (*C*) and to a conical expanding portion called the divergent (*E*) (*see* Fig. 1).

b) *Classical Venturi Tubes* — Venturi tube consists of a conical converging portion, convergent (*B*) connected to a cylindrical part called the throat (*C*) which in turn is connected to a conical expanding portion called the divergent (*E*). The venturi tube is a part of venturi meter (*see* Fig. 1).

c) *Venturi Nozzles* — If the convergent part of a venturi tube is a standard nozzle [*see* IS : 2952 (Part I)-1964* for details], the device is called a venturi nozzle (*see* Fig. 2).

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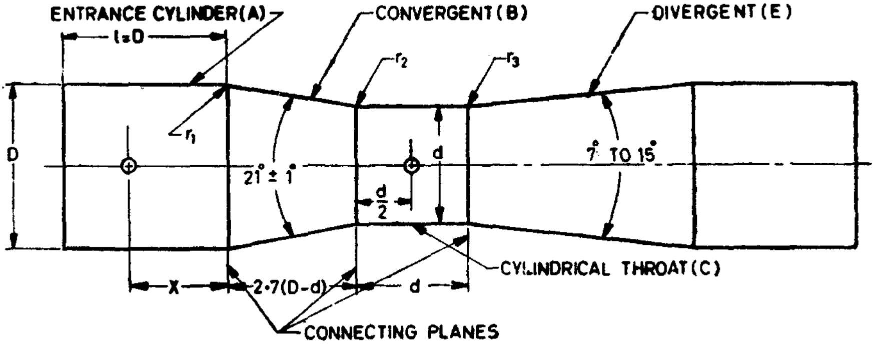
*Recommendation for methods of measurement of fluid flow by means of orifice plates and nozzles: Part I Incompressible fluids.



D = diameter of entrance cylinder (A)
d = diameter of cylindrical throat (C)
l = length of entrance cylinder (A)
r_1, r_2 and r_3 Radii

$$X = \frac{0{\cdot}25\ D \text{ to } 0{\cdot}75\ D \text{ for } 100 < D < 150 \text{ mm}}{0{\cdot}25\ D \text{ to } 0{\cdot}50\ D \text{ for } 150 < D < 800 \text{ mm}}$$

FIG. 1 LINE SKETCH OF A VENTURI METER

2.1.3 *Diameter Ratio* — The diameter of the throat of the primary element divided by the diameter of the pipe upstream of the primary element.

2.2 Pressure Measurement

2.2.1 *Pipe-Wall Pressure Tap* (*Piezometer Opening*) — A pipe-wall pressure tap is a hole drilled in the wall of a space breaking through the inside surface of the space and the edge of which is flush with the said surface. The hole is usually circular, but in certain cases may be of the annular slit type.

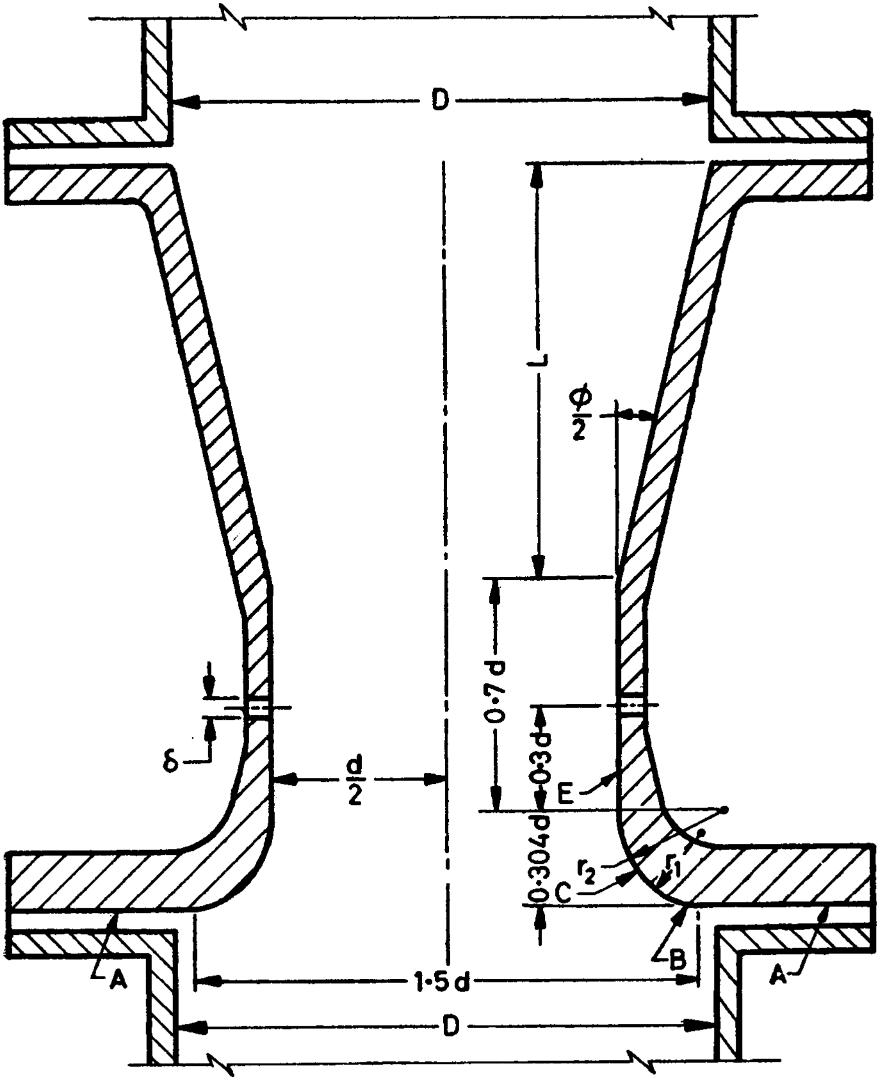
2.2.2 *Static Pressure* — The static pressure of a fluid flowing through a straight pipeline is the pressure which can be measured by connecting a pressure gauge to the pipe-wall pressure tap. Only the value of the absolute static pressure is considered in this standard.

2.2.3 *Differential Pressure* — Differential pressure is the difference between the static pressure which can be measured at pipe-wall taps (piezometer openings) at specified distances, one of which is on the upstream side and the other at the same level on the downstream side of a primary element inserted in a straight level pipe containing a flowing fluid.

NOTE — The term 'differential pressure' is used in this standard only if the pressure taps are in the positions specified for each standard primary element.

2.2.4 *Differential Pressure Ratio* — The differential pressure divided by the absolute static pressure existing at the level of the centre of the cross section of the pipe in the plane containing the centre-line of the upstream pressure tapping.

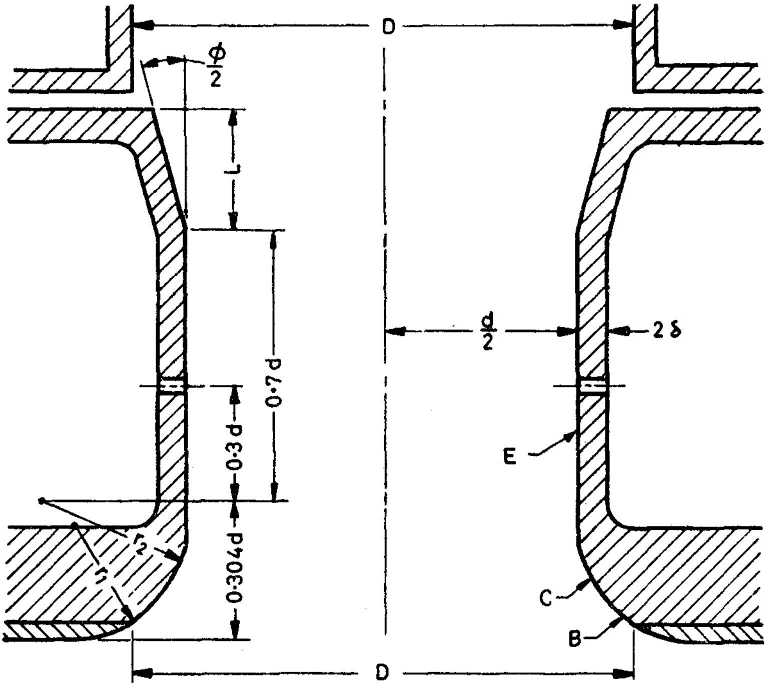
2.2.5 *Pressure Loss* — Difference in static pressure between the pressure measured on the upstream side of the primary element, at a point free from the influence of approach impact pressure, and that measured on the downstream side of the element, at a point where static pressure recovery by expansion of the jet is complete.



$d < 2/3\ D$

2A Non-truncated

FIG. 2 VENTURI NOZZLES (*Contd*)



d > 2/3*D*

2B Truncated

FIG. 2 VENTURI NOZZLES

2.3 Flow

2.3.1 *Rate of Flow* — The rate of flow of a fluid flowing through a primary element defines the quantity of fluid passing through the orifice or throat per unit time.

2.3.1.1 This quantity can be characterized by its mass or its volume and the flow value can be expressed in units of mass or volume per unit time.

2.3.1.2 In all cases it is necessary to state explicitly the type of flow value referred to, expressing either by mass or volume per unit time.

2.3.2 *Pipe Reynolds Number* — The pipe Reynolds number used in this standard is referred to the upstream condition of the fluid and to the upstream diameter of the pipe, that is:

$$Re_D = \frac{\bar{v}_1 D}{\nu_1}$$

where

Re_D = upstream Reynolds number related to D;
$\bar{v}_1$ = upstream mean velocity;
D = upstream diameter of the pipe; and
ν_1 = kinematic viscosity; applicable to the condition of the fluid in the pipe cross section containing the axis of the upstream pressure tap.

2.3.3 *Velocity of Approach Factor* — The velocity of approach factor is given by the equation:

$$E = (1-\beta^4)^{-\frac{1}{2}} = D^2/\sqrt{D^4-d^4} = (1-m^2)^{-\frac{1}{2}}$$

where

E = velocity of approach factor,
β = diameter ratio d/D,
D = upstream diameter of the pipe,
d = throat diameter, and
m = area ratio = β^2

2.3.4 *Flow Coefficients* — Experimental direct calibrations of standard primary element by means of incompressible fluids show that the value α defined by the following relation is only dependent on the Reynolds number for a given primary element in a given installation:

$$\alpha = \frac{q_m}{\frac{\pi}{4} d^2 \sqrt{2 \triangle P \rho_1}}$$

where

q_m = mass rate of flow,
d = throat diameter of the primary element at operating conditions,
$\triangle P$ = differential pressure, and
ρ_1 = mass density of fluid at the upstream section.

2.3.4.1 The value α, a pure number, is called the 'flow coefficient'. Its numerical value is the same for different installations whenever such installations are geometrically similar and flows are characterized by equal Reynolds numbers.

2.3.4.2 The ratio $C = \alpha/E$ is called the 'coefficient of discharge'.

2.3.5 *Expansibility (Expansion) Factor* — Calibration of a given primary element by means of a compressible fluid (gas), shows that the ratio:

$$\epsilon = \frac{q_m}{\alpha \frac{\pi}{4} d^2 \sqrt{2 \triangle P \rho_1}}$$

is dependent both on the value of the Reynolds number and on those of the differential pressure ratio and the isentropic exponent of the gas.

2.3.5.1 The method adopted for representing these variations consists in multiplying the flow coefficient α of the considered primary element, as determined by direct calibration effected by means of liquids for the same value of Reynolds number, by the 'expansibility', a so-called (expansion) factor defined by the relation:

$$\epsilon = \frac{q_m}{\frac{\pi}{4} d^2 \sqrt{2 \triangle P \rho_1}}$$

ϵ_1 differs from and is less than unity, when the fluid is compressible.

2.3.5.2 This method is possible because experiments show that practically ϵ_1 is independent of Reynolds number, and for a given diameter ratio of a given primary element, depends on the differential pressure ratio and the isentropic exponent.

2.3.6 *Isentropic Exponent (for the Compressible Fluid)* — The isentropic exponent κ appears, in the different formulae for expansibility (expansion) factor ϵ_1 either directly, or in the acoustic ratio 'X'. There are many gases and vapours for which no values for κ have been published so far. For gases, however, the behaviour of which fairly equals that of ideal gases, the isentropic exponent may be replaced by the ratio of the specific heat capacities. The isentropic exponent, as well as the ratio of the specific heat capacities, vary in general whenever the gas temperature and/or pressure vary.

2.3.7 *Acoustic Ratio* (X) — The differential pressure ratio divided by the isentropic exponent.

2.3.8 *Pressure Ratio* — The pressure ratio is the ratio of the absolute pressure at the throat taps to that at upstream taps.

3. PRINCIPLES OF THE METHOD OF MEASUREMENT

3.1 When a device, such as venturi meter or venturi nozzle is placed in a pipeline through which a fluid is flowing, a static pressure difference exists between the upstream side and the downstream side of the device and whenever the device is geometrically similar to one on which direct calibration has been made, the conditions of use being the same, the rate of flow may be determined from the measured value of this pressure difference and from a knowledge of the circumstances under which the device is being used.

4. SYMBOLS

4.1 The symbols used in this standard are given in Table 1.

5. GENERAL REQUIREMENTS FOR MEASUREMENT

5.0 All the requirements mentioned in this clause shall be simultaneously satisfied before and during the period of measurement.

TABLE 1 SYMBOLS

(*Clause* 4.1)

SYMBOL	REPRESENTED QUANTITY	DIMENSIONS*
α	Flow coefficient	Pure number
β	Diameter ratio, $\beta = \frac{d}{D}$	Pure number
C	Coefficient of discharge, $C = \frac{\alpha}{E}$	Pure number
E	Velocity of approach factor, $E = (1-\beta^4)^{-\frac{1}{2}}$	Pure number
ϵ	Expansibility (expansion) factor	Pure number
κ	Isentropic exponent†	Pure number
m	Area ratio, $m = \beta^2$	Pure number
R_{eD}	Reynolds number of upstream pipe referred to D	Pure number
x	Differential pressure ratio, $x = \frac{\Delta P}{P_1}$	Pure number
X	Acoustic ratio, $X = \frac{x}{\kappa}$	Pure number
d	Diameter of throat of primary element at operating conditions	L
D	Upstream pipe diameter at operating conditions	L
K	Absolute roughness	L
ΔP	Differential pressure	$ML^{-1}T^{-2}$
η	Dynamic viscosity of the fluid	$ML^{-1}T^{-1}$
ν	Kinematic viscosity of the fluid	L^2T^{-1}
P	Absolute static pressure of the fluid	$ML^{-1}T^{-2}$
q_m	Mass rate of flow	MT^{-1}
$q\bar{v}$	Volume rate of flow	L^3T^{-1}
ρ	Mass density of the fluid	ML^{-3}
t	Temperature of the fluid	—
$\bar{v}$	Mean axial velocity of the fluid in the pipe	LT^{-1}

*M = Mass. L = Length. T = Time.

†For ideal gases, the ratio of the specific heat capacities and the isentropic exponent have the same values.

Subscript ‘$^{-1}$’ applies to conditions (of the fluid, etc) in the plane of the upstream pressure tap.

Subscript ‘$^{-2}$’ applies to conditions (of the fluid, etc) in the plane of the downstream pressure tap.

5.1 Primary Element

5.1.1 The primary element shall be manufactured, installed and used in accordance with this standard.

5.1.2 The condition of the primary element shall be checked after each measurement or each series of such measurements or at sufficiently frequent intervals depending on the chemical properties of the gas to ensure that it conforms to the requirements of this standard. Accordingly the edges shall be invariably checked and any inequalities due to erosion or corrosion set right and the primary element calibrated before it is used again.

NOTE — The deposits or incrustations, if any, due to the chemical properties of the gas will cause variations in the coefficient of discharge in the course of time.

5.1.3 *Material* — The material used shall be non-rusting and non-corrodable and its coefficient of thermal expansion should be known. The primary element should be of stainless steel when cavitation or corrosion is expected.

5.2 Type of Gas — The gas shall be physically and thermally homogeneous and free from grit or suspended material above submicroscopic range (that is greater than 0·1 micron) and should be in the gaseous state throughout the period of measurement.

5.3 Installation Pipe System and Flow

5.3.1 *Installation*

5.3.1.1 The primary element is fitted between two sections of straight cylindrical pipe of constant cross-sectional area, in which there is no obstruction or branch connection (whether or not there is flow into or out of such connections during measurement) other than those specified in this standard. The pipe is considered straight when it appears so by mere visual inspection.

5.3.1.2 The minimum straight lengths of pipes to be installed upstream and downstream of the venturi meter for various types of disturbances are given in Table 2. For venturi nozzles, the values specified in IS : 2952 (Part I)-1964* may be considered as satisfactory.

5.3.1.3 When the straight lengths comply with the requirements of Table 2 and when they are longer than or equal to the values given for zero additional tolerance (values without bracket), it is not necessary to add any extra deviation to the flow measurement deviation to take into account an effect of such installation conditions.

5.3.1.4 When the upstream or downstream straight lengths are shorter than the zero additional tolerance values and greater than or equal to the '0·5 percent additional tolerance values' (bracketed values) as given in Table 2, an extra overall tolerance of 0·5 percent shall be added to the tolerance of the flow measurement.

5.3.1.5 The straight lengths of pipe specified in Table 2 are adequate under all normal conditions of pipe configurations to ensure that the flow at the plane of measurement does not contain any significant swirl. Swirling motion due to other kinds of upstream disturbances such as tangential or off-set entries, cyclones, vortex separators, certain partially-opened valves, etc, will require damping, since it can persist for long distances of the order of a hundred or more pipe diameters in unrestricted straight lengths of pipe.

Flow-straighteners of the honeycomb or nest-of-tubes type (*see* Fig. 3A) are very suitable where there is distortion or asymmetry of the flow and for mild swirling flow conditions. The maximum distance between the centres of the tubes should not exceed 0·25 times the pipe diameter and their overall length should be at least ten times this dimension.

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*Recommendation for methods of measurement of fluid flow by means of orifice plates and nozzles: Part I Incompressible fluids.

The arrangement shown in Fig. 3B has been found to destroy very severe swirling motion and is recommended for the types of upstream disturbances listed above. The perforations in the three plates should have a diameter equal to about half the plate thickness and shuold be uniformly distributed to occupy not more than half the pipe cross-sectional area.

Zanker flow-straightener shown in Fig. 3C gives a developed velocity profile in a short distance downstream.

The straightening device should be adequately robust and securely held in place within the pipe. It is usually most convenient to construct the straightener with a flange so that it can be fitted between two pipe flanges.

5.3.2 *Pipe System*

5.3.2.1 The pipe bore upstream of venturi tube should be circular and straight over the entire minimum lengths of straight pipe required (*see* Table 2). The cross section is taken to be circular if it appears so by mere visual inspection. The circularity of the outside of the pipe may be taken as a guide, except in the immediate vicinity of the primary element.

5.3.2.2 The mean diameter of the pipe *D*, where it joins the classical venturi tube should be within ±1 percent of the classical venturi tube entrance cylinder diameter. Moreover, no single diameter of this inlet pipe section should differ from the mean of the diameters by more than ± 2 percent for a distance of two pipe diameters preceding the classical venturi tube.

5.3.2.3 In case of venturi nozzle, over an upstream length of at least 2 *D* measured from the upstream face of the nozzle, the pipe should be cylindrical. The value of the diameter *D* of the pipe should be taken as the mean of the measurements of several diameters situated in meridian planes at approximately even angles to each other and in several planes normal to the pipe centre line within the specified length of 2 *D*. At least four diameter should be measured. The pipe is said to be cylindrical when no diameter in any plane differs by more than 0·3 percent from the value of *D* obtained as a mean of all measurements. Attention is called to the fact that it is possible to check circularity of a pipe bore, within the accuracy required, without measuring the mean diameter of the pipe bore itself.

5.3.2.4 The mean diameter of the pipe immediately downstream of venturi tube need not be measured accurately but it should be checked that, if the length of the divergent is less than four times the throat diameter, the downstream pipe diameter is not less than 90 percent of the diameter at the end of the venturi tube divergent. This means that, in most cases, pipes having the same nominal bore as that of the venturi tube may be used.

5.3.2.5 The inside diameter of the pipe should be at least 50 mm.

5.3.2.6 The inside surface of the measuring pipe immediately preceding the venturi tube should be clean, free from pitting and deposit and not encrusted. The pipe surface, at least for the minimum length given in Table 2, should have a roughness not greater than that of commercially smooth pipe (approximately $K < 0{\cdot}2$ mm).

5.3.3 *Flow*

5.3.3.1 The rate of flow should be constant or, in practice, vary only slightly and slowly with time. This standard does not provide for the measurement of pulsating flow.

5.3.3.2 The flow of the gas through the primary element should not undergo any change of phase (that is, from gaseous state to liquid). To determine whether there is a change in phase, the computation of flow should be carried out on the assumption that the expansion is isentropic and the ratio of the downstream to the upstream absolute pressures should be greater than 0·75.

6. COMPUTATION OF FLOW

6.1 Basic Formula

6.1.1 For calculating the mass rate of flow, q_m, the flow coefficient α and expansibility (expansion) factor ϵ, as specified in this standard, should be used in the following formula:

$$q_m = \alpha \, \epsilon \, \frac{\pi}{4} d^2 \sqrt{2 \, \Delta P \, \rho_1}$$

ϵ is equal to unity when the fluid is incompressible.

6.1.2 Similarly, the value of the volume rate of flow, at the upstream conditions of the fluid, may be calculated by the following relation:

$$q_v = q_m / \rho_1$$

6.1.3 The formulae of **6.1.1** and **6.1.2** apply for any consistent system of units.

6.2 Method of Determination of a Standard Primary Element — The principle of the method consists essentially in selecting a period — the type of standard primary element to be used, and a rate of flow and the corresponding value of the differential pressure.

6.2.1 The related values of q_m and ΔP should be inserted in the basic formula rewritten in the form below and the diameter ratio of the selected primary element is determined by successive approximations:

$$\alpha \beta^2 = \frac{4 q_m}{\epsilon \pi D^2 \sqrt{2 \, \Delta P \, \rho_1}}$$

6.3 Computation of Rate Flow — Computation of the rate of flow is effected by replacing the different terms on the right-hand side of the basic formula

$$q_m = \alpha \, \epsilon \, \frac{\pi}{4} d^2 \sqrt{2 \Delta P \rho_1}$$

TABLE 2 MINIMUM STRAIGHT LENGTHS OF PIPE REQUIRED BETWEEN VARIOUS FITTINGS LOCATED UPSTREAM AND DOWNSTREAM OF THE VENTURI METER AND THE DEVICE ITSELF

(*Clauses* 5.3.1.2, 5.3.1.3, 5.3.1.4, 5.3.2.1, 5.3.2.6 *and* 9.3.1)

DIAMETER RATIO	GATE VALVE FULLY OPEN	SINGLE 90° SHORT RADIUS ELBOWS*	TWO OR MORE 90° BENDS IN THE SAME PLANE*	TWO OR MORE 90° BENDS IN DIFFERENT PLANES†, *	TWO OR MORE 90° BENDS IN DIFFERENT PLANES WITH STRAIGHTENING VANES*	REDUCTION 3 *D* TO *D* IN A LENGTH OF 3·5 *D*	INCREASE 3/4 *D* TO *D* IN A LENGTH OF *D*	DOWN-STREAM (ANY KIND OF FITTING)
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)
0·40	2·5 (1·5)	0·5 (‡)	1·5 (0·5)	(0·5)	3·5 (1)	2·5 (0·5)	1·5 (0·5)	4 *D*
0·45	3·5 (1·5)	1·0 (0·5)	1·5 (0·5)	(0·5)	3·5 (1)	4·5 (0·5)	2·5 (1)	,,
0·50	3·5 (1·5)	1·5 (0·5)	2·5 (1·5)	(8·5)	4·5 (0·5)	5·5 (0·5)	2·5 (1·5)	,,
0·55	4·5 (2·5)	2·5 (0·5)	2·5 (1·5)	(12·5)	5·5 (0·5)	6·5 (0·5)	3·5 (1·5)	,,
0·60	4·5 (2·5)	3·0 (1·0)	3·5 (2·5)	(17·5)	5·5 (0·5)	8·5 (0·5)	3·5 (1·5)	,,
0·65	4·5 (2·5)	4·0 (1·5)	4·5 (2·5)	(23·5)	5·5 (0·5)	9·5 (1·5)	4·5 (2·5)	,,
0·70	5·5 (3·5)	4·0 (2·0)	4·5 (2·5)	(27·5)	5·5 (3·0)	10·5 (2·5)	5·5 (3·5)	,,
0·75	5·5 (3·5)	4·5 (3·0)	4·5 (3·5)	(29·5)	6·5 (3·5)	11·5 (3·5)	6·5 (4·5)	,,

Fittings	*Minimum Inlet Straight Length Required*
Thermometer pockets of diameter $\leqslant 0·03\,D$	5 (3)
Thermometer pockets of diameter between 0·03 and 0·13 *D*	20 (10)

NOTE 1 — The straight lengths presented in this table are minimum values expressed as multiples of *D*, and it is always recommended to have straight lengths larger than those indicated. Especially, for research work it is recommended to at least double the upstream values given in this table 'for zero additional tolerance values'.

NOTE 2 — This table is valid only for venturi meters, as defined in this standard, having diameter ratios between 0·4 and 0·75.

NOTE 3 — The unbracketed values are 'zero additional tolerance values'. The bracketed values are '0·5 percent additional tolerance values'.

NOTE 4 — This standard does not give any information, if the straight lengths are shorter than the corresponding bracketed values. This is also the case when the upstream and downstream straight lengths are simultaneously shorter than 'zero additional tolerance values'.

NOTE 5 — The valves mentioned in this table should be fully open. It is recommended to make the control, if necessary, by valves located downstream from the primary element. The isolating valves located upstream will be preferably of the 'gate valve' type and fully open.

NOTE 6 — The following remarks may be used as a guide in cases of practical installation:

a) If the primary element is installed in a pipe leading from an open space or a large vessel, either directly, or through any fittings given in this table the total length of the pipe between the open space and the primary element shall never be less than 30 *D* (as in the case of orifice plates and nozzles).

b) If several fittings other than 90° elbows, are placed in succession upstream from the primary element, the following rule should be applied:

> Between the closest fitting (first) to the primary element and the primary element itself, keep a minimum straight length, such as indicated for fitting (first) in question and the actual value in this table. But, in addition, keep between the fitting (first) and the β preceding one (second) a straight length equal to one-half of the value given in this table for fitting (second) and for the value of $\beta = 0·7$ (whatever the actual value of β may be). This rule is not applicable when the fitting (second) is an abrupt symmetrical reduction, which case is covered in the paragraph (a) above.

The radius of curvature of the elbow (short radius bend) must be equal to or greater than the pipe diameter.

As the effect may still be present after 40 *D*, no unbracketed values can be given in the table.

‡Since no fitting can be placed closer than 0·5 *D* to the upstream pressure tappings of the venturi tube the zero additional tolerance is applicable in this instance.

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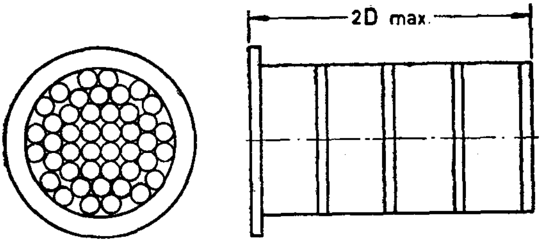
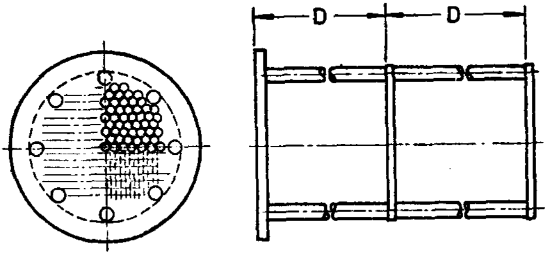
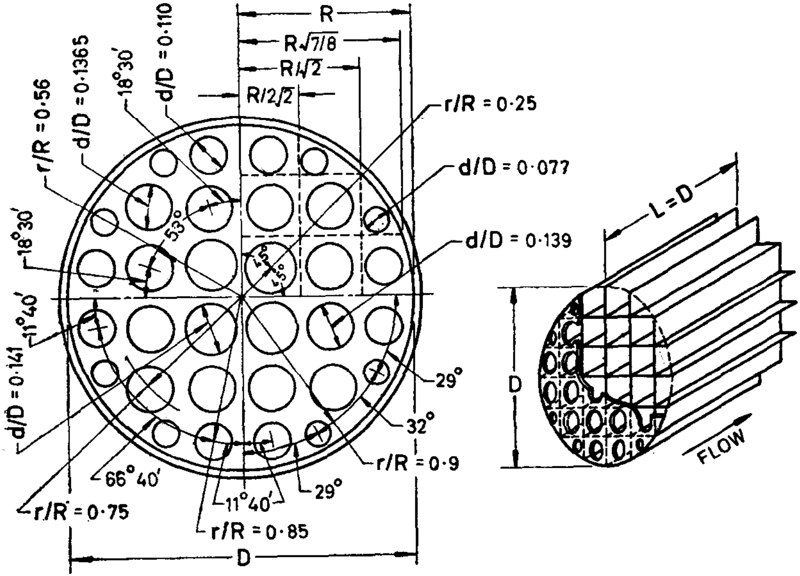


3A NEST OF TUBES



3B PERFORATED PLATES



3C THE ZANKER FLOW-STRAIGHTENER

FIG. 3 DETAILS OF STRAIGHTENING VANES

by their numerical values, obtained in the course of the measurement, and by calculating their product. The computation itself involves no difficulty other than of an arithmetical nature and merely calls for the following comments:

a) α may be dependent on R_{eD} which is itself dependent on q_m. Therefore, the final value of q_m may be obtained by successive approximations, after first calculating q_m from a value of R_{eD} (or of α) chosen a priori. For instance, $\alpha = \alpha_0$ can be taken as a first value in case of venturi nozzle.

b) ΔP represents the differential pressure, as defined in **2.2.3**.

NOTE 1 — The corrected diameter 'α' is obtained from the measured diameter 'd_m' according to the equation:

$$d = d_m \, [1 + \theta \ \ (t - t_m)]$$

where θ is the mean coefficient of linear expansion per degree Centigrade of the material forming the orifice or throat, t is the temperature (°C) of the device when in use and t_m is the temperature (°C) at which the diameter was measured.

NOTE 2 — For easy comparison, it is preferable to convert the quantity to a standard temperature and pressure (for example 0°C, 760 mm of mercury).

7. CLASSICAL VENTURI METERS

7.0 Field of Application — Classical venturi meters cause a much smaller pressure loss than standard orifice plates or nozzles with the same area ratio. Their range of use, within the compass of this standard, depends on the way they are manufactured, as indicated in **7.0.1**, **7.0.2** and **7.0.3**.

7.0.1 Classical venturi meters with a rough-cast convergent may be used in pipes in the range of diameters between 100 and 800 mm provided the diameter ratio β lies between 0·3 and 0·75 inclusive ($0{\cdot}09 \leqslant \beta^2 \leqslant 0{\cdot}56$).

7.0.2 Classical venturi meters with a machined convergent may be used in pipes in the range of diameters between 50 and 250 mm provided the diameter ratio β lies between 0·4 and 0·75 inclusive ($0{\cdot}16 \leqslant \beta^2 \leqslant 0{\cdot}56$).

7.0.3 Classical venturi meters with a rough welded sheet-iron convergent may be used in pipes in the range of diameters between 200 and 1 200 mm provided the diameter ratio β lies between 0·4 and 0·7 inclusive ($0{\cdot}16 \leqslant \beta^2 \leqslant 0{\cdot}49$).

7.1 Geometric Profile

7.1.1 Figure 1 gives a section passing through the axis of the throat of a classical venturi meter.

7.1.1.1 The internal surface of the device shall be circular and coaxial with the pipe axis. The venturi meter shall be made of a cylindrical portion (A), connected to a convergent (B), a cylindrical throat (C), and a conical divergent (E). Simple visual inspection is sufficient to ascertain the coaxiality of the convergent and the cylindrical portion of the throat.

7.1.2 *Dimensional Requirements*

7.1.2.1 *Entrance cylinder* — The cylinder *A* shall have an entrance diameter *D* which shall not differ from the pipeline inside diameter by more than ±0·01 *D*. The minimum length of the entrance cylinder, measured from the plane containing the intersection of the convergent *B* with cylinder *A* should be equal to 1 *D*, but may vary as a result of the manufacturing processes (*see* **7.2**). The entrance diameter *D* shall be measured very carefully in the plane of the upstream pressure tappings, near each pair of pressure tappings and also midway between these pairs of tappings. The number of measurements shall be equal to the number of pressure tappings. The mean value of all the measurements shall be taken as the value of *D*. No diameter along the cylindrical portion shall differ by more than ± 0·4 percent from the value of the mean diameter.

7.1.2.2 *Convergent* — The convergent *B* shall be conical and should have a total included angle of 21±1° for any type of classical venturi tube. Its upstream limit is the plane containing the intersection of the convergent with the entrance cylinder (or their expansion) and downstream limit is the plane containing the intersection of the convergent with the throat *C* (or their extensions). The overall length of the convergent parallel to the axis shall therefore be approximately equal to 2·7 $(D-d)$. The profile of the convergent portion shall be checked with a template. The deviation between the template and the truncated section of the convergent should not exceed 0·4 percent of *D* in any place. The junction of the convergent *B* and the cylinder *A* should have a curvature of radius r_1, the value of which depends on the type of the venturi meter (*see* **7.2**). Rounding off the radius r_1 may be checked with a template.

7.1.2.3 *Throat* — The throat *C* should be cylindrical, diameter *d*. Its limits extend from the plane containing the intersection of the convergent with the throat (or their extensions) to the plane containing the intersection of the throat with the divergent *E* (or their extensions). The length of throat *C*, equal to the distance between these two planes, shall be equal to *d* for all types of classical venturi tube. The throat *C* should be connected to the convergent *B* by a curvature of radius r_2 and to the divergent *E* by a curvature of radius r_3. The values of r_2 and r_3 would depend upon the type of classical venturi tube (*see* **7.2**). The values of the radii of curvature r_2 and r_3 should be checked with a template. The maximum value of the deviation should not exceed 0·02 *d*. The diameter *d* shall be measured very carefully at the section of the throat pressure tappings. The number of measurements should be at least the same as the number of pressure tappings. The diameter shall be measured near each pair of the pressure tappings and also at the centre of these pairs of tappings. The mean value of all these measurements shall be taken as the value of *d*. No diameter along the throat shall differ by more than 0·1 percent of the value of the mean diameter. The roughness of the throat and that of the adjacent radii should be less than $10^{-5}d$.

7.1.2.4 *Divergent* — The divergent E shall be conical and may have an included angle lying between 7° and 15°; it is, however, preferable to have an angle between 7° and 8°. The divergent may be truncated down to about 35 percent of its length without significant modification of the pressure loss in the device. This allows a reduction in divergent length without the angle being modified.

NOTE — A venturi tube is called truncated when the outside diameter of the divergent is less than the diameter D and not truncated when the diameter is equal to D.

The internal surface of the truncated section shall be axi-symmetric and it may be considered so when two diameters situated in the same plane perpendicular to the axis do not differ from the value of the mean diameter by more than 0·4 percent.

7.1.2.5 *The roughness criterion* (R_a)* of the throat and that of the adjacent curvature should be less than $10^{-5}d$ and if possible equal to $0{\cdot}5 \times 10^{-5}d$. The divergent is rough-cast. Its internal surface should be clean and smooth. The roughness of other parts of the classical venturi tube depends on the type considered.

7.2 Manufacture

7.2.0 There are three methods of manufacturing venturi meters resulting in the meters having slightly different flow characteristics. These methods are characterized by the method of manufacture of inside surface of convergent portion and the amount of rounding at the intersection of the convergent portion and the throat. They are the following:

a) Meters made by casting in a sand mould or by other methods of fabrication leaving a finish on the surface of the convergent portion similar to that of the sand casting, with the throat and the entrance cylinder being machined and also the junctions between the cylinders and cones rounded;

b) Meters cast or fabricated as above but in which the convergent portion, the throat and the inlet cylinder are all machined. They may or may not be rounded at the junctions; and

c) Meters manufactured by welding and which are not finished in any way in the larger sizes, or with only the throat finished, in smaller sizes.

Venturi tubes of sizes up to 150 mm may preferably be cast in one piece and for larger sizes, the castings should not be more than in three sections, namely, convergent, throat and divergent.

Parts of the venturi meter may be fitted by sweating on or with flanges. In the case of fitting by sweating process there should not be any relative movement between parts under their normal conditions of working. Flanges, if used, shall conform to IS : 1538-1969†. The inside surface shall be smoothly finished without any pits or protuberances. The end connection of the venturi meter shall be flanged and the size and holes in the flanges should conform to IS : 1538-1969†.

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*Arithmetical mean deviation R_a from the central line of the profile.

†Specification for cast iron fittings for pressure pipes for water, gas and sewage (*first revision*).

7.2.1 *Characteristics of the Profile of the Venturi Meters with a Rough Cast Convergent* — The minimum length of cylinder *A* may be equal to the smaller of the following two quantities:

$$1D \text{ or } 0{\cdot}25\ D+250 \text{ mm}$$

7.2.1.1 The radius of curvature r_1 should be equal to 1·375 D(±20 percent). The radius of curvature r_2 should be equal to 3·625 $d(\pm 0{\cdot}125\ d)$.

7.2.1.2 The length of the throat shall be greater than $d/3$ and the length lying between the end of the joining curvature r_2 and the plane of pressure tappings shall not be less than $d/6$. The radius of curvature r_3 should lie between 5 *d* and 15 *d*. Its value increases as the divergent angle decreases. A value close to 10 *d* is recommended for a divergent included angle close to 7°.

7.2.1.3 Irrespective of the radius r_3 adopted, the length of the throat lying between the plane of the downstream pressure tappings and the beginning of the curvature of the radius r_3 shall not be less than $d/6$.

7.2.1.4 The internal surface of the convergent is sand cast and rough cast. It should be free from cracks and fissures, depressions, irregularities and impurities. The roughness should be less than $10^{-4}D$.

7.2.1.5 The internal surface of cylinder *A* may also remain rough-cast if it shows the same surface finish as that of the convergent.

7.2.2 *Characteristics of the Profile of the Venturi Meters with a Finished Convergent* — The minimum length of the cylinder *A* shall be equal to 1*D*, the radius of curvature r_1 should be less than 0·25 *D* and preferably equal to zero, the radius of curvature r_2 shall be less than 0·25 *d* and preferably equal to zero. The length of the throat lying between the end of the curvature r_2 and the plane of the pressure tappings to the throat shall not be less than 0·25 *d*. The radius of curvature r_3 should be less than 0·25 *d* and preferably equal to zero. The length of the throat lying between the connection and the plane of the pressure tappings to the throat should not be less than 0·3 *d*. The entrance cylinder and the convergent should have a surface finish similar to that of the throat.

7.2.3 *Characteristics of the Profile of the Venturi Meters with a Rough Welded Sheet Iron Convergent* — The minimum length of the cylinder *A* should be equal to 1 *D*. There should not be joining curvature between cylinder *A* and convergent *B*, between convergent *B* and throat *C* and the throat and the divergent, other than that resulting from the welding. The internal surface of cylinder *A* and convergent *B* should be clean, not encrusted and free from welding deposits. It may be galvanized. Its roughness should be less than $5\times10^{-4}\ D$. The height of the internal welded seams should not exceed 5 percent of the diameter at the plane of weld.

8. VENTURI NOZZLES

8.1 A venturi nozzle (*see* Fig. 2A and 2B) consists of a convergent portion, a cylindrical throat of diameter *d* and a divergent.

8.1.1 *Convergent* — When d is less than $2/3\ D$ (Fig. 2A), the plane A is limited by a circumference whose diameter is $1{\cdot}5\ d$, and by the inside perimeter of the pipe of diameter D. When $d=2/3\ D$, there is no plane A.

8.1.1.1 When d is more than $2/3\ D$, (Fig. 2B), the nozzle is manufactured as if d is less than $2/3\ D$ and the inlet flat part is then faced off so that its largest diameter is just equal to D.

8.1.1.2 When d is smaller than or equal to $2/3\ D$, the arc of circumference B should be tangential to the flat inlet part A.

8.1.1.3 The radius r_1 should be equal to $0{\cdot}2\ d \pm 10$ percent for diameter ratio (β) less than 0·5 and $0{\cdot}2\ d \pm 3$ percent for diameter ratios (β) greater than or equal to 0·5. The centre of r_1 should be at $0{\cdot}2\ d$ from the inlet plane and at $0{\cdot}75\ d$ from the centre line.

8.1.1.4 The arc of circumference C should be tangential to the arc of circumference B and to the throat E.

8.1.1.5 The radius r_2 should be equal to $d/3 \pm 10$ percent for β less than 0·5 and to $d/3 \pm 3$ percent for β greater than 0·5. The centre of r_2 should be at 5/6 d from the axial central line and at $0{\cdot}304\ d$ from the flat inlet part A.

8.1.2 *Throat* — The throat E having a diameter d should have a length $0{\cdot}7\ d$. The value d of the diameter of the throat should be taken as the mean of the measurements of at least four diameters situated in meridian planes distributed at approximately 45° to each other. No two throat diameters should differ from one another by more than $0{\cdot}001\ d$. The throat diameter d may vary between 50 mm and approximately 390 mm.

8.1.3 *Divergent* — The divergent should be connected with the throat without a rounded part, but any burrs should be removed. The total angle of the divergent should be $\phi \leqslant 30°$. The length L of the divergent has practically no influence on the flow coefficient α. The included angle of the divergent and hence the length, however, does influence, the pressure loss.

8.2 The internal surface of all parts of the venturi nozzle should have a roughness less than $10^{-5}\ d$.

9. PRESSURE TAPPINGS AND TEMPERATURE MEASUREMENT

9.0 General Location of Pressure Tappings — The axes of the pressure tappings may be located in any meridian half plane of the pipe. The axis of the upstream tapping and that of the downstream tapping may be located in different meridian half planes. However, differential pressure should be computed in accordance with **2.2.3**.

NOTE — It is preferable to ensure that the tappings are at the top.

9.1 Pressure Tappings for Classical Venturi Tubes

9.1.1 The pressure tappings at any section should be made in the form of not less than four individual wall tappings inter-connected by an annular chamber. The diameter of these tappings, should in general be between 6 mm and 12 mm but in no case greater than 0·1 *D* for the upstream and 0·13 *d* for the throat pressure tappings. The centre lines of the pressure tappings should be equally spaced along the circumference and should be contained in a plane perpendicular to the axis of venturi meter.

9.1.1.1 The bore of the pressure tapping hole should be generally circular. The edges should be flush with the pipe-wall, free from burrs and show no ridges nor, in general, any irregularity and if rounding off is required the radius should not exceed 1/10 of the diameter of pressure tapping. Unless otherwise specified, the pressure tappings should be an even number.

9.1.1.2 The conformity of the pressure tappings with the above descriptions may be assessed by visual inspection.

9.1.2 The distance of the upstream pressure tapping, measured along a straight line parallel to the axis of the venturi tube, from the plane containing the intersection of the entrance cylinder (*A*) and the convergent (*B*) should be as follows:

a) For venturi meters with a rough cast convergent, between 0·25 *D* and 0·75 *D* for *D* between 100 and 150 mm and between 0·25 *D* and 0·50 *D* for *D* between 150 and 800 mm, and
b) For venturi meters with a finished convergent and with a rough welded sheet-iron convergent, $0{\cdot}5D \pm 0{\cdot}05\ D$.

9.1.3 For all types of venturi meters, the plane of the axes of the throat pressure tappings should be at a distance of 0·5 *d* (±0·02 *d*), from the plane containing the intersection of convergent *B* and throat *C*.

9.1.4 The annular chamber of the pressure tappings should be such that the area of the free cross section of annular chamber is greater than or equal to half the total surface area of the openings of the pressure tappings connecting the chamber to the entrance cylinder or the throat (*see* Fig. 4A).

9.2 Pressure Tappings for Venturi Nozzles

9.2.1 The upstream pressure tappings should be located at the junction of the upstream pipe and the convergent and should conform to the following.

9.2.1.1 The pressure tappings should be either not less than four individual tappings as in the case of venturi meters or more usually annular slits opening into annular chambers of piezometric rings as shown in Fig. 4B.

9.2.1.2 The diameter of individual tappings or the width of annular slit δ should be as given below:

For $\beta \leqslant 0{\cdot}65$, $\delta = 0{\cdot}03\ D$, and

For $\beta > 0{\cdot}65$ δ should be between 0·01 *D* and 0·02 *D*.

But in no case δ should be less than 1 mm and greater than 10 mm.

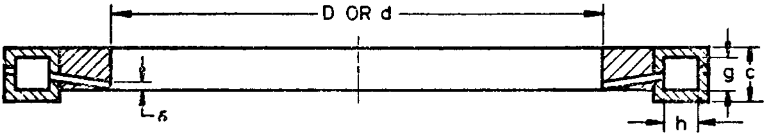
9.2.2 Throat tappings should be individual pressure tappings, leading into an annular chamber of the piezometer ring. The pressure tapping should be located in a plane at a distance of 0·3 *d* from the upstream end of the throat.

9.2.2.1 Individual pressure tappings should be at least four in number, their axes should be at equal angles to one another and lie in a plane perpendicular to the centre line of the venturi nozzle.

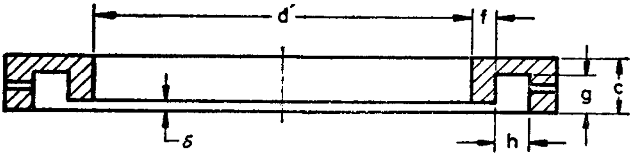
9.2.2.2 The tappings should always be wide enough to ensure that choking by dirt plugs, or gas bubbles is prevented. The diameter of individual tappings in the throat of venturi nozzles should be between 2 mm and 10 mm and they should be less than 0·04 *d*.

9.2.3 *Piezometric Ring* — The inner diameter *d′* of the upstream piezometer ring (*see* Fig. 4B) should lie between *D* and 1·02 *D*, and its axial length (*C*) should be smaller than 0·2 *D*. It shall be noted that the diameter ratio β should be calculated using the pipe diameter *D* and not diameter *d′* of the ring.

9.2.3.1 The thickness of the annular slit *f* (or the length of the single tap) should be equal to or more than twice the width δ of the slit (or of the diameter of the single tap). The area of the free cross section of the annular chamber ($g \times h$) should be equal to or more than half the total area of the tapping holes connecting the chamber to the pipe.



4A INDIVIDUAL TAPPINGS



4B ANNULAR SLITS OPENING INTO THE ANNULAR CHAMBER OF PIEZOMETRIC RINGS

FIG. 4 PRESSURE TAPPINGS

9.2.3.2 All surfaces of the ring in contact with the liquids should be cleaned and machined.

9.2.3.3 The pressure tappings connecting the annular chambers to the secondary devices should be pipe wall tappings. They should be circular with diameters between 4 mm and 10 mm.

9.3 Temperature Measurement

9.3.1 The temperature of the fluid should be measured preferably upstream of the primary element and the temperature measured nearest to 0·5°C. The thermometer pocket should take up as little space as possible.

The distance between it and the venturi meter shall be in accordance with the values specified in Table 2.

9.3.2 Any method of determining the static pressure and the temperature of the fluid is acceptable if it permits of obtaining a reliable value of the temperature, the viscosity and the mass density of the fluid in the section being metered upstream of the primary element without disturbing the flow in the area of measurement. The temperature of the primary element is assumed to that of the fluid entering it.

9.4 Thermal Insulation

9.4.1 The metering section of the pipe shall be lagged for at least the whole length of the required straight runs. The flanges surrounding the primary element should be lagged.

9.4.2 It is, however, unnecessary to lag the pipe when the temperature of the fluid between the inlet of the minimum straight length of the metered pipe on the upstream side and the outlet of the straight length on the downstream side, does not vary beyond such value as the user may determine according to the accuracy he requires in the flow measurement.

10. NOMINAL SIZES

10.1 The nominal sizes shall refer to the nominal internal diameter of the pipeline.

10.2 Venturi meters and venturi nozzles shall be made of the following nominal sizes:

> 50, 65, 80, 100, 150, 200, 250, 300, 350, 400, 450, 500, 600, 700, 750, 800, 900, 1 000 and 1 200 mm.

10.3 The minimum wall thickness and the design of venturi meter and venturi nozzle shall be such that they provide strength and rigidity equivalent to a pipe of corresponding nominal size conveying the same liquid under the same pressure, conforming to the relevant Indian Standard for pipes.

11. MATERIALS FOR MANUFACTURE

11.1 The venturi meters or nozzles may be manufactured of any material and in any manner whatsoever, provided that they conform and continue to conform to the provisions given in this standard throughout the period of their use.

11.2 Recommended Materials

11.2.1 *Cast Iron* — Grade 20 of IS : 210-1970* with minimum Brinell's hardness of 80 when tested with a 10 mm diameter ball under a load of 1 000 kg applied for 15 seconds.

11.2.2 *Gun Metal* — Gun metal is recommended when greater consistency of results is required over an extended period of time. The gun metal shall be of quality not less than that specified in IS : 306-1968* with an elongation of not less than 12 percent on a gauge length of 5 cm when tested in accordance with IS : 497-1953†. The minimum Brinell hardness shall be 80 when tested with a 10 mm diameter ball under a load of 1 000 kg applied for 15 seconds.

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*Specification for grey iron castings.

11.3 Other Material for Use — Other materials may be used if necessary, provided the following conditions are satisfied:

a) Facility and convenience to machine the parts within the tolerance specified,
b) Relatively low coefficient of thermal expansion,
c) No chemical or injurious action on the wall by the flowing fluid, and
d) Strength to maintain the shape without any alteration under all conditions of flow.

11.4 Packing Material — The packing material between flanges, if any, may be of thin rubber or packing paper or card board and should not protrude inside and obstruct the passage of the fluid.

12. COATING

12.1 In the case of cast iron, immediately after machining, a thin protective coat of paint shall be given both inside and outside.

12.2 The coating shall not affect the internal diameter and shall not be affected by the flowing liquid.

13. MARKING

13.1 Each venturi meter (also venturi nozzle) shall have a name-plate bearing the following details:

a) The manufacturer's name or trade-mark, and
b) The size of venturi meter that is both the inlet and the throat size.

13.1.1 Venturi meters (and also venturi nozzles) may also be marked with the ISI Certification Mark.

NOTE — The use of the ISI Certification Mark is governed by the provisions of the Indian Standards Institution (Certification Marks) Act and the Rules and Regulations made thereunder. The ISI Mark on products covered by an Indian Standard conveys the assurance that they have been produced to comply with the requirements of that standard under a well-defined system of inspection, testing and quality control which is devised and supervised by ISI and operated by the producer. The ISI marked products are also continuously checked by ISI for conformity to that standard as a further safeguard. Details of conditions, under which a licence for the use of the ISI Certification Mark may be granted to manufacturers or processors, may be obtained from the Indian Standards Institution.

14. DISCHARGE COEFFICIENTS

14.1 Classical Venturi Tubes

14.1.1 *Range of Application*

14.1.1.0 For the application of the coefficients given in this standard, the limitations given in **14.1.1.1** to **14.1.1.3** should be taken into consideration.

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*Specification for tin bronze ingots and castings (*revised*).

†Specification for tensile testing of metals (non-ferrous).

14.1.1.1 Venturi tubes with a 'rough-cast' convergent may be used in pipes with diameters between 100 and 800 mm provided the diameter ratio lies between 0·3 and 0·75 inclusive ($0{\cdot}09 \leqslant m \leqslant 0{\cdot}56$).

14.1.1.2 Venturi tubes with a finished convergent may be used in pipes with diameters between 50 and 250 mm provided the diameter ratio lies between 0·4 and 0·75 inclusive ($0{\cdot}16 \leqslant m \leqslant 0{\cdot}56$).

14.1.1.3 Venturi tubes with a rough welded iron sheet convergent may be used in pipes with diameters between 200 and 1 200 mm provided the diameter ratio lies between 0·4 and 0·7 inclusive ($0{\cdot}16 \leqslant m \leqslant 0{\cdot}5$).

14.1.2 *Setting Up of Venturi Tube*

14.1.2.1 *Upstream pipe system* — When the upstream straight length exceeds 15 D, the pipe system should be smooth so as to enable the use of the flow coefficient given in **14.1.3**. In all cases the roughness of the upstream pipe should be less than 0·000 1 D on a length at least equal to 2 D counted upstream from the venturi tube.

14.1.2.2 *Drain holes* — The pipe system may be fitted with drain holes necessary for the preliminary evacuation of solid deposits and fluids other than the measured fluid. There should be no flow through these drain holes while flow measurement is in progress.

14.1.3 *Discharge Coefficients*

14.1.3.1 *Limits of use* — Whatever may be the type of venturi tubes a simultaneous use of extreme values for D, β and Re_D should be avoided (*see* Note under **5.1.2**).

14.1.3.2 For installations outside the limits defined in **14.1.3.3**, **14.1.3.4** and **14.1.3.5** for D, β or Re_D it is essential to calibrate separately the primary element in its actual conditions of service. The effects of Re_D, K/D, β and C are still insufficiently known to be able to give reliable values of C outside the limits defined in **14.1.3.3** to **14.1.3.5** for each type of venturi tube. .

14.1.3.3 *Discharge coefficient of the classical venturi tube with a rough cast convergent* — The value of discharge coefficient is given as $C=0{\cdot}984$ within the following limits:

$$100 \text{ mm} \leqslant D \leqslant 800 \text{ mm}$$
$$0{\cdot}3 \leqslant \beta \leqslant 0{\cdot}75$$
$$2 \times 10^5 \leqslant Re_D \leqslant 2 \times 10^6$$

14.1.3.4 *Discharge coefficient of the venturi tubes with finished convergent* — The value of discharge coefficient is given as $C=0{\cdot}995$ within the following limits:

$$50 \text{ mm} \leqslant D \leqslant 250 \text{ mm}$$
$$0{\cdot}4 \text{ mm} \leqslant \beta \leqslant 0{\cdot}70 \text{ mm}$$
$$2 \times 10^5 \leqslant Re_D \leqslant 1 \times 10^6$$

14.1.3.5 *Discharge coefficient of the venturi tube with a rough welded iron convergent* — The value of the discharge coefficient is given as $C=0{\cdot}985$ within the following limits:

$$200 \text{ mm} \leqslant D \leqslant 1\,200 \text{ mm}$$

$$0{\cdot}4 \leqslant \beta \leqslant 0{\cdot}7$$
$$2 \times 10^5 \leqslant Re_D \leqslant 2 \times 10^6$$

14.2 Venturi Nozzles

14.2.1 *Range of Application* — These may be used in pipes with diameter varying from 65 to 500 mm and for area ratios varying from 0·1 to 0·6. The lower limit for D is dependent upon the area ratio and the diameter of the throat tapping.

14.2.2 *Setting Up of the Venturi Nozzle* (*Drain Holes*) — The pipeline should be provided with the necessary drains for the removal of solid deposits and liquids. These drain holes shall not be used while measurement is in progress. Only in unavoidable cases, the drain holes may be placed near the venturi nozzle, in which case the diameter of these drain holes should be smaller than 0·08 D, and their location should be such that the distance measured on a straight line from one of the holes to a pressure tapping on the same side of the venturi nozzle always exceeds 0·5 D. Furthermore the axis of the drain holes should be situated radially so that it does not interfere with any pressure tap, when the pressure taps used are not annular pressure taps.

14.2.3 *Flow Coefficients*

14.2.3.1 The flow coefficients α_0 for venturi nozzles in smooth pipes are dependent upon the area ratio m (*see* Table 3). In Table 3 m^2 instead of m is chosen because it permits a linear interpolation of α_0. Extrapolation is never permissible.

TABLE 3 FLOW COEFFICIENTS α_0 AS A FUNCTION OF m^2 FOR VENTURI NOZZLES

m	m^2	α_0	m	m^2	α_0
0·1	0·01	0·989	0·435 9	0·19	1·061
0·141 4	0·02	0·993	0·447 2	0·20	1·066
0·173 2	0·03	0·997	0·458 3	0·21	1·070
0·200 0	0·04	1·001	0·469 0	0·22	1·075
0·223 6	0·05	1·005	0·479 6	0·23	1·080
0·244 9	0·06	1·009	0·439 9	0·24	1·086
0·264 6	0·07	1·013	0·500 0	0·25	1·091
0·282 8	0·08	1·016	0·509 9	0·26	1·096
0·300 0	0·09	1·020	0·519 6	0·27	1·102
			0·529 2	0·28	1·107
0·316 2	0·10	1·024	0·538 5	0·29	1·113
0·331 7	0·11	1·027			
0·346 4	0·12	1·031	0·547 7	0·30	1·119
0·360 6	0·13	1·035	0·556 8	0·31	1·124
0·374 2	0·14	1·039	0·565 7	0·32	1·130
0·387 3	0·15	1·043	0·574 5	0·33	1·136
0·400 0	0·16	1·048	0·583 1	0·34	1·142
0·412 3	0·17	1·052	0·591 6	0·35	1·149
0·424 3	0·18	1·057	0·600 0	0·36	1·155

14.2.3.2 The flow coefficient $\alpha = \alpha_0 \times r_0$ in rough pipes is the product of α_0 (as contained in Table 3) and the correction factor r_0 for pipe roughness as contained in Table 4. Guide values for the mean roughness K in mm of pipe walls are given in Table 5.

TABLE 4 VALUES OF r_0

D/K β^2	400	800	1 200	1 600	2 000	2 400	2 800	3 200
	Values of r_0							
0·3	1·002	1·000	1·000	1·000	1·000	1·000	1·000	1·000
0·4	1·003	1·002	1·000	1·000	1·000	1·000	1·000	1·000
0·5	1·008	1·005	1·003	1·002	1·001	1·000	1·000	1·000
0·6	1·014	1·009	1·006	1·004	1·002	1·001	1·000	1·000
0·65	1·016	1·012	1·009	1·007	1·005	1·003	1·002	1·000

TABLE 5 GUIDE VALUES FOR THE ROUGHNESS K OF PIPE WALLS

Sl No. (1)	Material (2)	Condition (3)	K in mm (4)
i)	Brass, copper, aluminium, plastic, glass	Smooth, without sediments	<0·03
ii)	Steel	New, seamless cold drawn	<0·03
		New, seamless warm drawn	0·05 to 0·10
		New, seamless rolled	
		New, welded longitudinally	
		New, welded spirally	0·10
		Slight rusted	0·10 to 0·20
		Rusty	0·20 to 0·30
		Encrusted	0·50 to 2
		With heavy incrustations	>2
		Bituminized, new	0·03 to 0·05
		Bituminized, normal	0·10 to 0·20
		Galvanized	0·13
iii)	Cast iron	New	0·25
		Rusty	1·01 to 1·5
		Incrusted	>1·5
		Bituminized, new	0·1 to 0·15
iv)	Asbestos cement	Insulated and not insulated, new	<0·03
		Not insulated, normal	0·05

14.2.3.3 Figure 5 shows the lower and upper limits of Reynolds number for which the flow coefficient may be taken as constant with good approximation, and Table 6 may also be used for convenience.

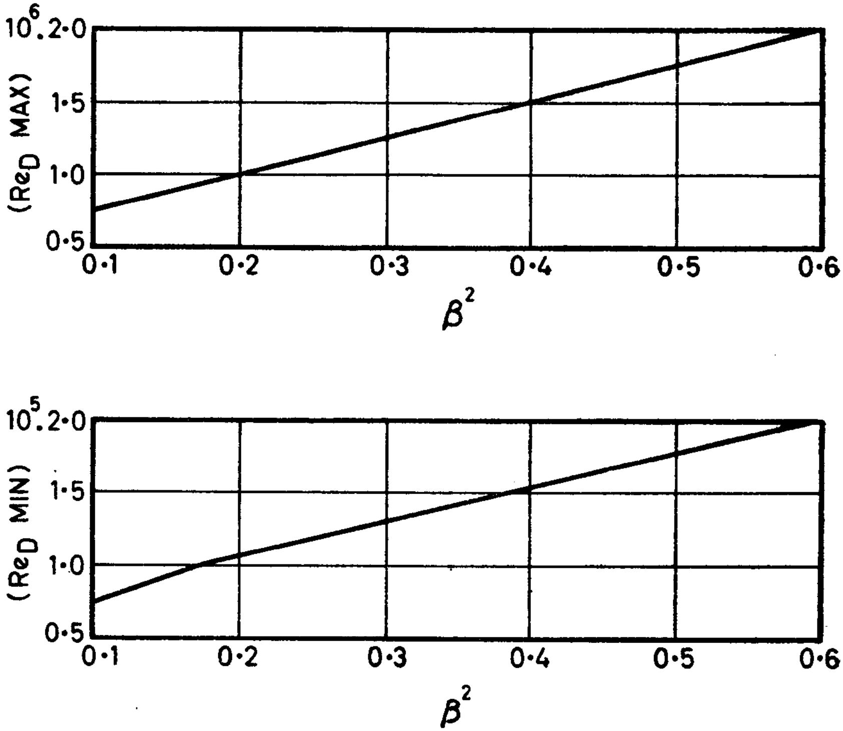

FIG. 5 VENTURI NOZZLE—MAXIMUM AND MINIMUM LIMITS OF REYNOLDS NUMBER R_{e_D} FOR THE STANDARD FLOW COEFFICIENT

14.3 Expansibility (Expansion) Factor — The theoretical equation for the expansibility factor for venturi tubes and nozzles is as follows:

$$\epsilon = \left[\left(\frac{\kappa \lambda^{\frac{2}{\kappa}}}{\kappa-1}\right) \left(\frac{1-\beta^4}{1-\beta^4 \lambda^{\frac{2}{\kappa}}}\right) \left(\frac{1-\lambda^{\frac{\kappa-1}{\kappa}}}{1-\lambda}\right) \right]^{\frac{1}{2}}$$

where $\lambda = (1-x)$

14.3.1 Values of the expansibility factor can be read from Table 7 (or alternately by Fig. 6), for a range of isentropic exponents, pressure ratios and area ratio values. Figure 7 gives the details of specific heat ratios which has the same numerical value as the isentropic exponent.

14.3.2 In particular, it should be noted that there may be increasing differences between the actual and the theoretical conditions when the pressure ratio decreases below 0·75.

TABLE 6 LIMITING VALUES OF REYNOLDS NUMBER Re_D FOR FLOW COEFFICIENTS OF VENTURI NOZZLES

(*Clause* 14.2.3.3)

β^2	β^4	Re_D	
		Min ($\times 10^5$)	*Max* ($\times 10^6$)
(1)	(2)	(3)	(4)
0·100 0	0·01	0·75$10^5$	0·75$10^6$
0·141 4	0·02	0·90$10^5$	0·85$10^6$
0·173 2	0·03	1·00$10^5$	0·93$10^6$
0·200 0	0·04	1·08$10^5$	1·00$10^6$
0·223 6	0·05	1·13$10^5$	1·06$10^6$
0·244 9	0·06	1·18$10^5$	1·11$10^6$
0·264 6	0·07	1·23$10^5$	1·16$10^6$
0·282 8	0·08	1·27$10^5$	1·21$10^6$
0·300 0	0·09	1·31$10^5$	1·25$10^6$
0·316 2	0·10	1·35$10^5$	1·29$10^6$
0·331 7	0·11	1·38$10^5$	1·33$10^6$
0·346 4	0·12	1·42$10^5$	1·37$10^6$
0·360 6	0·13	1·45$10^5$	1·40$10^6$
0·374 2	0·14	1·48$10^5$	1·43$10^6$
0·387 3	0·15	1·51$10^5$	1·47$10^6$
0·400 0	0·16	1·54$10^5$	1·50$10^6$
0·412 3	0·17	1·57$10^5$	1·53$10^6$
0·424 3	0·18	1·60$10^5$	1·56$10^6$
0·435 9	0·19	1·62$10^5$	1·59$10^6$
0·447 2	0·20	1·65$10^5$	1·62$10^6$
0·458 3	0·21	1·67$10^5$	1·64$10^6$
0·469 0	0·22	1·70$10^5$	1·67$10^6$
0·479 6	0·23	1·72$10^5$	1·70$10^6$
0·489 9	0·24	1·75$10^5$	1·72$10^6$
0·500 0	0·25	1·77$10^5$	1·75$10^6$
0·509 9	0·26	1·79$10^5$	1·77$10^6$
0·519 6	0·27	1·81$10^5$	1·80$10^6$
0·529 2	0·28	1·84$10^5$	1·82$10^6$
0·538 5	0·29	1·86$10^5$	1·84$10^6$
0·547 7	0·30	1·88$10^5$	1·87$10^6$
0·556 8	0·31	1·90$10^5$	1·89$10^6$
0·565 7	0·32	1·92$10^5$	1·91$10^6$
0·574 5	0·33	1·94$10^5$	1·93$10^6$
0·583 1	0·34	1·96$10^5$	1·96$10^6$
0·591 6	0·35	1·98$10^5$	1·98$10^6$
0·600 0	0·36	2·00$10^5$	2·00$10^6$

---

15. OVERALL ACCURACY

15.1 If the measurements have been carried out in good conditions and according to this standard, and the flow rate during the time taken for the measurement has changed only slowly and in any case the variations have not exceeded ±5 percent of the reference flow rate, the overall accuracy of the measurement is estimated to be within ±1 percent of the true mean flow rate over the period of measurement.

TABLE 7 EXPANSIBILITY (EXPANSION) FACTOR FOR VENTURI TUBES FOR GASES OR STEAM IN TERMS OF THE PRESSURE RATIO $\frac{P_2}{P_1}$, OF β^4, AND OF THE ISENTROPIC EXPONENT κ*

(*Clause* 14.3.1)

β^2	β^4 \ $\frac{P_2}{P_1}$	1·0	0·98	0·96	0·94	0·92	0·90	0·85	0·80	0·75
					ϵ for $\kappa=1·2$					
0	0	1·0	0·987 4	0·974 8	0·962 0	0·949 1	0·936 1	0·902 9	0·868 9	0·834 0
0·316 2	0·1	1·0	0·985 6	0·971 2	0·956 8	0·942 3	0·927 8	0·891 3	0·854 3	0·816 9
0·447 2	0·2	1·0	0·983 4	0·966 9	0·950 4	0·934 1	0·917 8	0·877 3	0·837 1	0·797 0
0·547 7	0·3	1·0	0·980 5	0·961 3	0·942 4	0·923 8	0·905 3	0·860 2	0·816 3	0·773 3
0·632 5	0·4	1·0	0·976 7	0·954 1	0·932 0	0·910 5	0·889 5	0·839 0	0·790 9	0·744 8
0·640 3	0·41	1·0	0·976 3	0·953 2	0·930 8	0·909 0	0·887 7	0·836 6	0·788 1	0·741 6
					ϵ for $\kappa=1·3$					
0	0	1·0	0·988 4	0·976 7	0·964 9	0·952 9	0·940 8	0·910 0	0·878 3	0·845 7
0·316 2	0·1	1·0	0·986 7	0·973 4	0·960 0	0·946 6	0·933 1	0·899 0	0·864 5	0·829 4
0·447 2	0·2	1·0	0·984 6	0·969 3	0·954 1	0·939 9	0·923 7	0·885 9	0·848 1	0·810 2
0·547 7	0·3	1·0	0·982 0	0·964 2	0·946 6	0·929 2	0·912 0	0·869 7	0·828 3	0·787 5
0·632 5	0·4	1·0	0·978 5	0·957 5	0·936 9	0·916 8	0·897 1	0·849 5	0·803 9	0·759 9
0·640 3	0·41	1·0	0·978 1	0·956 7	0·935 8	0·915 4	0·895 4	0·847 2	0·801 2	0·756 9
					ϵ for $\kappa=1·4$					
0	0	1·0	0·989 2	0·978 3	0·967 3	0·956 2	0·944 9	0·916 2	0·886 5	0·855 8
0·316 2	0·1	1·0	0·987 7	0·975 3	0·962 8	0·950 3	0·937 7	0·905 8	0·873 3	0·840 2
0·447 2	0·2	1·0	0·985 7	0·971 5	0·957 5	0·943 0	0·928 8	0·893 3	0·857 7	0·821 9
0·547 7	0·3	1·0	0·983 3	0·966 7	0·950 3	0·934 0	0·917 8	0·878 0	0·838 8	0·800 0
0·632 5	0·4	1·0	0·980 0	0·960 4	0·941 2	0·922 3	0·903 8	0·858 8	0·815 4	0·773 3
0·640 3	0·41	1·0	0·979 6	0·959 6	0·940 1	0·920 9	0·902 1	0·856 6	0·812 7	0·770 4
					ϵ for $\kappa=1·66$					
0	0	1·0	0·990 9	0·981 7	0·972 4	0·962 9	0·953 3	0·928 8	0·903 3	0·876 8
0·316 2	0·1	1·0	0·989 6	0·979 1	0·968 5	0·957 8	0·947 1	0·919 7	0·891 7	0·862 9
0·447 2	0·2	1·0	0·987 9	0·975 9	0·963 7	0·951 6	0·939 4	0·908 8	0·877 8	0·846 4
0·547 7	0·3	1·0	0·985 8	0·971 8	0·957 7	0·943 8	0·929 9	0·895 3	0·860 9	0·826 5
0·632 5	0·4	1·0	0·983 1	0·966 4	0·949 9	0·933 6	0·917 6	0·878 2	0·839 7	0·802 0
0·640 3	0·41	1·0	0·982 7	0·965 7	0·949 0	0·932 4	0·916 1	0·876 2	0·837 3	0·799 3

*The values for $\beta^2 = \beta^4 = 0$ and $\frac{P_2}{P_1}=1$ are given in order to make it possible to interpolate for values of ϵ for $\beta^4<0·1$ and $\frac{P_2}{P_1}>0·98$. The fact that values are given to four figures does not indicate the accuracy of ϵ.

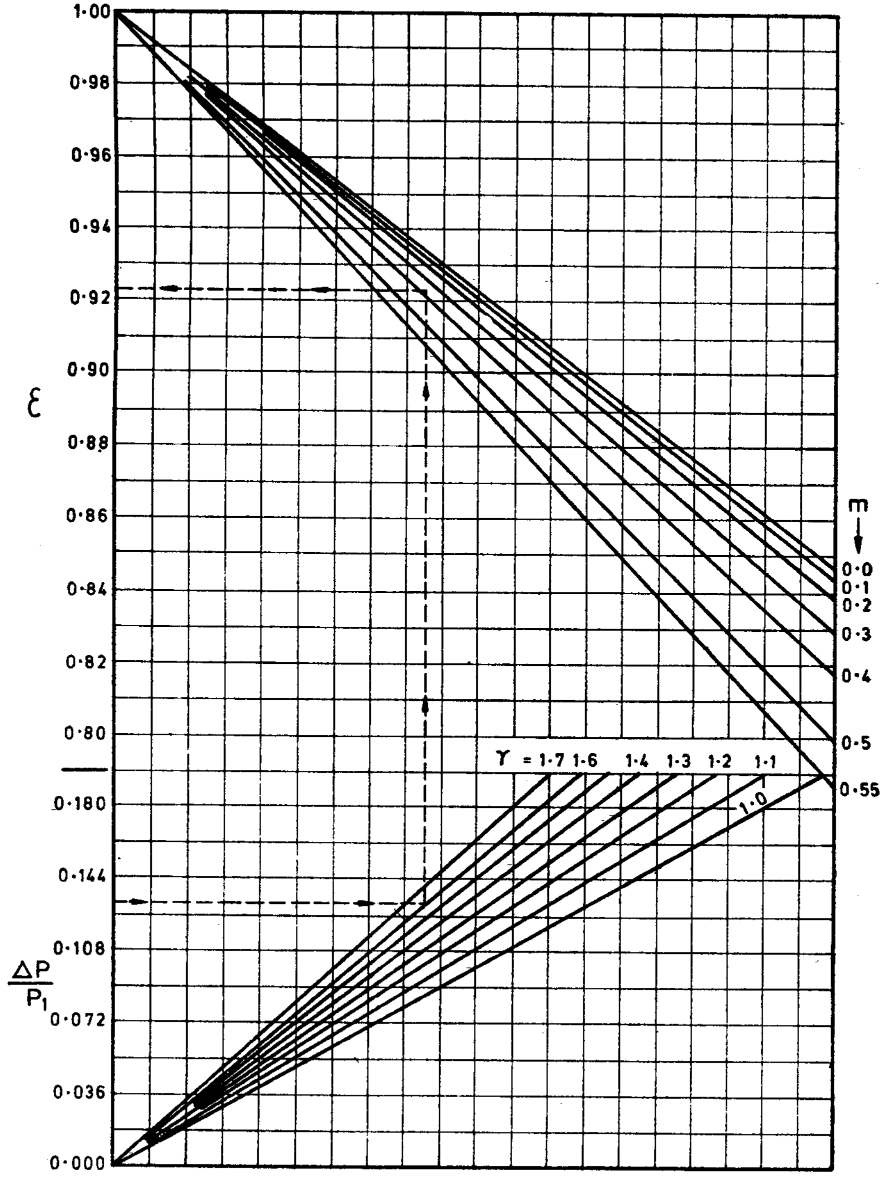

FIG. 6 EXPANSIBILITY FACTORS FOR VENTURI METERS AND VENTURI NOZZLES

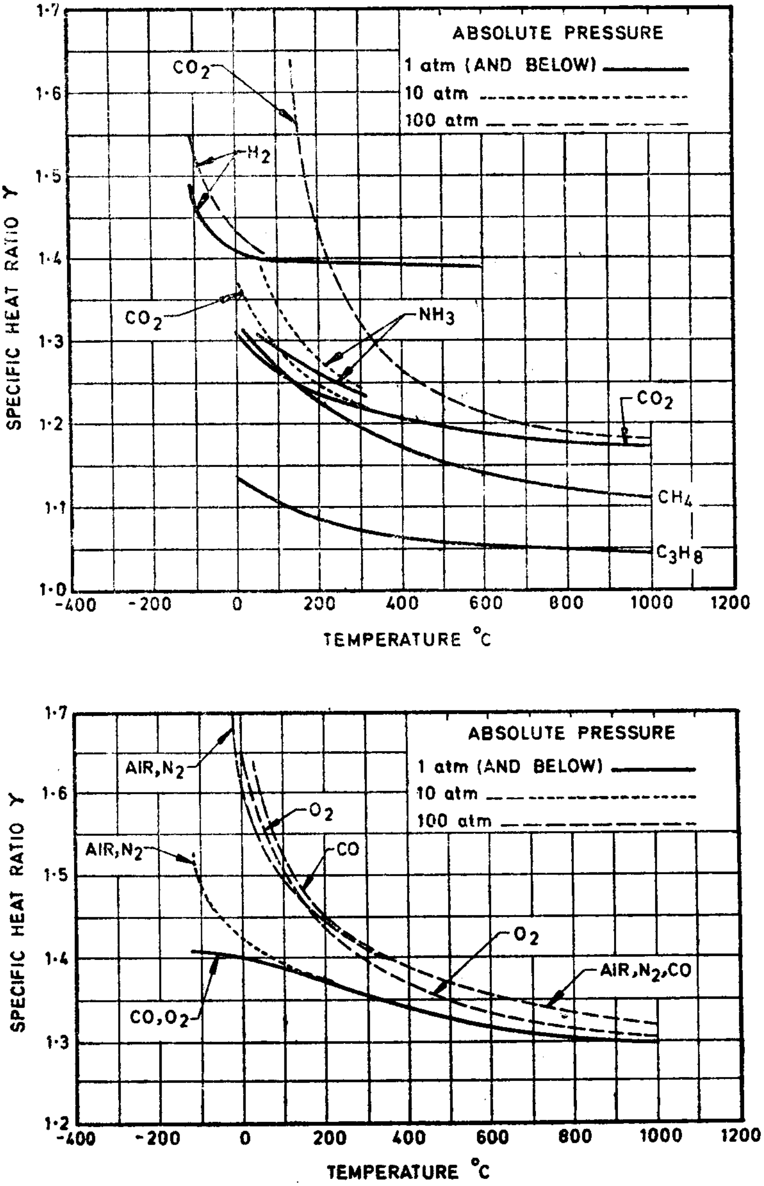

FIG. 7 SPECIFIC HEAT RATIOS OF SELECTED GASES

APPENDIX A

(*Clause* 1.4)

EXAMPLE FOR CALCULATION OF FLOW RATE

A-1. MASS RATE OF FLOW

A-1.1 Find mass rate of flow under the following conditions:

a) Measurement temperature 0°C
b) $D=0{\cdot}05$ m
c) $\beta=0{\cdot}6$, that is $d=0{\cdot}03$
d) Type — venturi meter with rough cast convergent
e) $P_1=10$ atmospheres = 1 013 373 N/m^2
f) $P_2=8$ atmospheres = 810 698 N/m^2
g) Type of gas — oxygen
h) Density (at the upstream pressure and temperature) = 14·29 kg/m^3
j) Machined convergent

A-2. PROCEDURE

A-2.1 The basic formula is:

$$q_m = \epsilon . \alpha . \frac{\pi}{4} d^2 \sqrt{2 \triangle P \rho_1}$$

ϵ depends on κ the isentropic exponent which has got the same value as γ the specific heat ratio. From Fig. 7 γ works out to 1·66 which is the value of κ. Referring to Table 7, for $\beta^4=0{\cdot}129\,6$, $\frac{P_1}{P_2}=0{\cdot}8$ and $\kappa=1{\cdot}66$, we get ϵ as 0·887 4.

$$\therefore \quad q_m = \alpha \times 0{\cdot}887\,4 \times \frac{\pi}{4} \times 0{\cdot}03 \times 0{\cdot}03 \sqrt{2 \times 202675 \times 14{\cdot}29}$$

$$= 1{\cdot}510\,\alpha$$

$$\alpha = CE = C(1-\beta^4)^{\frac{1}{2}}$$

$$C = 0{\cdot}984 \text{ (vide } \mathbf{14.1.3.3}\text{)}$$

$$\therefore \quad \alpha = 1{\cdot}055 \text{ and}$$

$$q_m = 1{\cdot}593 \text{ kg/s}$$

$$Re_D = \frac{q_m \quad D}{\pi/4 \times D^2 \times \rho \times \gamma}$$

Assuming kinematic viscosity as $0{\cdot}135 \times 10^{-5}$ m^2/s

$$Re_D = \frac{1{\cdot}593 \times 4 \times 7 \times 10^5}{22 \times 0{\cdot}05 \times 14{\cdot}29 \times 0{\cdot}135} = 2{\cdot}07 \times 10^6$$

which is marginally outside the limits but not likely to cause significant change in the results.

(*Continued from page* 2)

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